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The thermal characteristics of a helical coil heat exchanger for seawater-source heat pump in cold winter

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Abstract

A high-density polyethylene helical coil heat exchanger (HCHE) is firstly adopted by a seawater-source heat pump system (SWHP) and experiments are conducted to study the thermal performance of the heat exchanger in icy condition. The external convective heat transfer coefficient is calculated by the experimental results and used in the development of the mathematical models. To predict the heat transfer process of the HCHE in icy condition in frigid periods of winter, a mathematical model is developed. Simulations with different parameters are conducted to investigate the effects of variable parameters on the thermal performance, such as inlet temperature, intermediate medium's flow rate, heat exchanger's length and diameter, temperature of seawater. The study indicates that the developments of the mathematical model are very helpful in the designing of heat exchangers used for the SWHP system.

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Keywords: Helical coil heat exchanger; Heat transfer coefficient; Icy condition; Numerical simulation

1. Introduction

According to a survey by the ASHRAE Handbook [1] and the design data of global companies in Europe and the United States, research has focused on closed-loop rather than open-loop systems, for the open-loop system has lower energy efficiency and may not operate when the pipe is corroded and the seawater is frozen. Compared with the open loop SWHP system, the closed loop system can avoid these phenomena due to indirect contact between the unit and seawater, and it requires lower energy consumption, as it does not need to overcome the resistance of the seawater transport process [2,3].

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The experiment was conducted in Bohai Sea in Tianjin, North China. As a semi-closed inland sea, the Bohai Sea is east of China, specifically, surrounded by Liaoning, Hebei, Shandong and Tianjin provinces. Affected by monsoons, the climate of the Bohai Gulf is cold and dry in winter and damp and warm in summer. This study monitored the coastal seawater temperature and speed of the Bohai Sea in winter, and the typical temperature distribution of winter are shown in Fig.1. Influenced by cold air in winter, the surface seawater temperature was usually below 0 °C, leading to the freezing of the seawater surface. Because the average salinity of the Bohai Sea is 3‰, the freezing point of seawater in this area is -1.7 °C. In general, the icy period of the Bohai Sea lasts 2 to 3months, starting in late November or early December.

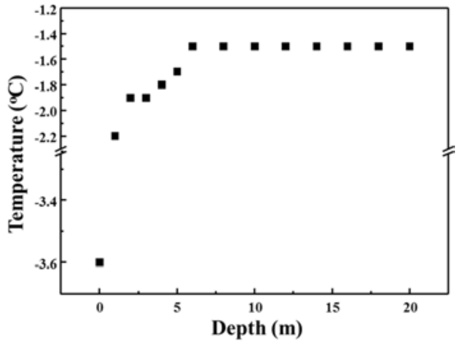

Fig. 1 The typical seawater temperature distribution of Bohai Gulf in winter

Fig. 2 The seawater velocity vector on the surface layer

Bohai Gulf is a semi-closed gulf, and the seawater velocity in the inner bay is merely 50% of that in the outer bay. From the monitoring records, a typical velocity distribution of the seawater was presented in Fig.2. The heat exchanger was settled at spot 1#. Beside spot 1#, the monitoring of water velocity was also conducted at spot 2#, 3# and 4#, where were around 1# to get all-sided accurate information. According to the longterm monitored records, the seawater velocity on the surface layer varied from 0.15 to 0.64m/s, and the velocity changed from 0.12 to 0.34m/s, 0.07 to 0.22m/s when it was measured at the depth of 5m and 10m undersea, respectively.

As illustrated by Stig [4] that subsea heat exchangers studied before are all based solely on natural convection. As a follow-up study of Yu [5], field tests were carried out in the sea to study the heat transfer characteristics of the HCHE in heating condition. For the rapidly changing of the seawater flow rate, there is no equations can describe the heat transfer process of the helical coil heat exchanger precisely. The external convective heat transfer coefficients of the exchanger were calculated by the experimental data. Mathematical models have been developed to describe the heat transfer process of HCHE using the external convective heat transfer coefficient calculated by experimental results. The effects of inlet temperature, intermediate medium’s flow rate, heat exchanger’s length and diameter, temperature of seawater on the heat transfer performance were simulated in cooling condition in icy condition in winter.

Nomenclature

C_{in}	specific heat, J/kg·K
D	curvature radius of the HCHE, m
d_{icei}	diameter of the ice layer outside the pipes, m
d_{in}	internal diameter of the pipes, m
d_{out}	external diameter of pipes, m
h_{ice}	external convective heat transfer coefficient, W/ (m^2·K)
h_{in}	internal convective heat transfer coefficient, W/ (m^2·K)
t_{icei}	temperature of the ice outside the pipes, °C
t_{ini}	temperature of the pipes' inner wall, °C
t_{fi}	temperature of the intermediate medium, °C
t_{sea}	seawater temperature, °C
t_{wi}	temperature of the pipes' outer wall, °C
u_{in}	velocity of the intermediate medium, m/s
ρ_{in}	density of the intermediate medium, kg/m^3
λ_{ice}	heat conductivity coefficient of the ice layer, W/(m·K)

2. Experimental setup

Weight and size of subsea heat exchangers are crucial design parameters. The exchanger should be designed to minimize both variables without compromising quality, safety and reliability. Because of their inaccessibility on the seabed, subsea heat exchangers must be robust and not require frequent maintenance. Therefore, a new type of helical coil heat exchanger (HCHE) is developed in the present study. The exchanger is composed of high density polyethylene pipes with 100 m long and 40 mm in diameter. The thickness of the pipes is 8mm. The pipes are coiled around the fixed mount, which is welded by stainless steel and treated with preservatives to avoid corrosion from seawater.

To fulfil the demands of the heat transfer area and to limit the total height, the heat exchanger is designed as a spiral, which has two staggered circles, one inside and another outside. The diameters of the two circles are 0.9 m and 1.3 m respectively, as shown in Fig.3. This design solves the issue of immobilizing the heat exchanger at the bottom of the sea. According to the experimental results of the seawater temperature in winter, the seawater temperature across a depth of 5 m to 20 m could reach -1.5 °C in the coldest period, which is close to the freezing point of seawater. Hence, the HCHE would freeze when it works. To avoid the adhesion of two icy layers nearby, the vertical space between the two pipes is set to 140 mm. Thus, both the inner circle and the outer circle are coiled 14 laps, yielding a fixed mount measuring 1.8 m tall.

At the bottom of the fixed mount, a 0.5m-high base is set in case the pipes sink into the mud. Rocks are used to fortify the base to prevent the heat exchanger from shaking while immersed in the sea.

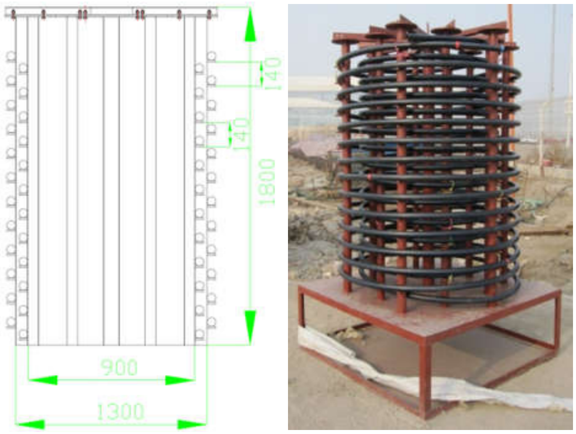

Fig.3 Helical coil heat exchanger

The SWHP system has three different fluid loops, one for the refrigerant in the heat pump and two for the intermediate medium, which comprises the water-ethylene glycol mixture used for transporting energy from and to seawater and the water used to transport energy from and to fan coil. The schematic diagram of the SWHP system is shown in Fig. 4. The heating-cooling changeover is achieved by changing the direction of the intermediate medium using four-way valves. In the heating mode, the intermediate medium heated in the seawater is directed to the evaporator, where it rejects heat to the refrigerant. The intermediate medium follows the path marked with H. The transport water heated in the condenser is directed to the indoor fan-coil, where it rejects heat to the conditioned room air. The path followed by the transport water is also marked with H. The pipes between the HCHE and the heat pump unit are 15 m long.

To make the fine adjustment of the water-inlet temperature, a thermostat with K-type thermocouples is positioned at the entrance of the HCHE. Water temperatures along the HCHE are measured using T-type copper-constantan thermocouples inserted into the pipes every ten meters from the entrance. Insulation to ensure water-tight conditions is implemented at the measure points. Thermal flow meters with Pt100 temperature sensors and ultrasonic flow meters are used to monitor the temperature differences and the flow rate between the inlet and outlet of the condenser and the evaporator. The temperatures of the seawater and the outdoor air are monitored with T-type copper-constantan thermocouples. The pressure differences between the inlet and the outlet of the HCHE are tested with a pressure transducer. The temperature test is performed automatically with T-type copper-constantan thermocouples using a Fluke Data Acquisition device. All thermocouples used are checked in a high-precision constant temperature water bath. The accuracy of the instruments used in the experiment is specified in Table 1.

Table 1 Accuracy of the instruments

Instruments	Accuracy	Instruments	Accuracy
T-type copper-constantan thermocouples	±0.5 °C	Pt100 temperature sensors	$\pm(0.15+0.002\,|t|)$ ℃
Pressure transducer	±1%	Ultrasonic flow meter	±2%

An important issue is the accuracy of the measured data and experimental results. An uncertainty analysis was conducted to validate the experimental results. According to Moffat [6], the uncertainty of the temperature distribution along the HCHE is established to be ±2.06%. The uncertainty of the intermediate medium flow rate in the HCHE is ±3.79%. The uncertainty of the convective heat transfer coefficients between the HCHE and the seawater is approximately ±3.12%. The heat transfer rate associated with the temperate and the flow rate is calculated to be ±4.57%, an acceptable uncertain value range in engineering application.

Monitoring data of seawater temperature in Tianjin coastal areas indicate that the lowest seawater temperature in winter can be -1.5°C. A water-ethylene glycol mixture with a mass fraction of 25% and a freezing point of -13°C are selected as the heat transfer medium to prevent the fluid in the HCHE from freezing.

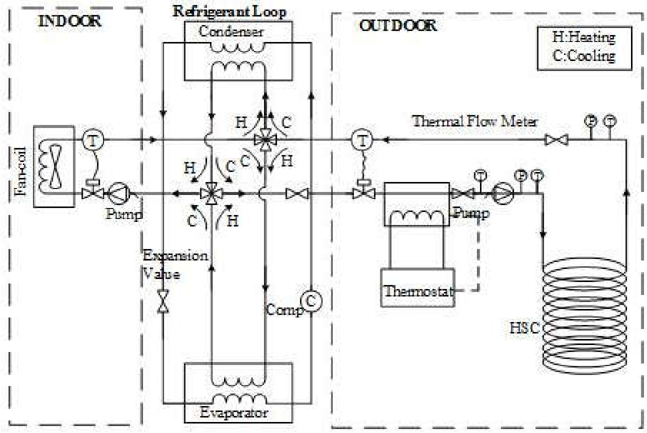

Fig.4 Schematic diagram of the SWHP system

3. Test results

Experiments were conducted to study the heat transfer characteristics of the HCHE during frigid periods in winter. The outdoor air and the seawater temperature were approximately -4°C and -1.5°C, respectively. The intermediate medium’s flow rate in the HCHE was set at 0.2m/s and the inlet temperature was -7.5°C. The heat exchanger was placed into the seawater 5 m away from the shore, where the maximum seawater depth is 14 m when the tide rises while the minimum depth is 11 m when the tide falls. Fig.5 gives the ice distribution along the HCHE. The heat exchanger was lift out of the seawater while the system ran smoothly. The ice thickness was measured by vernier caliper. The entire subsea heat exchanger was covered with ice. From Fig.6 we could see that the ice outside the pipes decreases from the inlet to the outlet. The maximum diameter of ice layer achieved outside the pipes was 95 mm which happened at the surface of the seawater, while the diameter of ice layer outside the pipes at the outlet of the HCHE was approximately 41.5 mm during frigid periods in winter, as shown in Fig. 6.

Fig.5 Icing along the HCHE

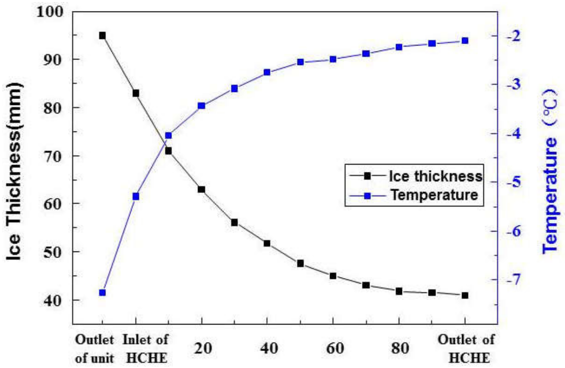

Fig.6 Diameter of the ice layer outside the pipes and the temperature of the water-ethylene glycol mixture distribution along the HCHE in icy conditions in winter

4. Mathematical modeling

There is no empirical formula to calculate the external convective heat transfer coefficients with respect to the uncertainty of the seawater velocity in magnitude and direction. Therefore, the external convective heat transfer coefficients are experimentally determined to provide a reference for future studies.

The external convective heat transfer coefficients were calculated using the methods of forward elimination and backward substitution. The intermediate medium temperature every 10 m along the pipes at different flow rate, inlet temperature and ice thickness were tested in the experiments. The test results were used to solve the external convective heat transfer coefficients. The external convective heat transfer coefficients every 10 m were calculated with the methods of forward elimination and backward substitution to ensure the accuracy of the calculation. The entire heat transfer process seemed to remain in steady-state. The schematic diagram of the heat transfer in icy condition was shown in Fig.7. The calculation process was specified by the following equations:

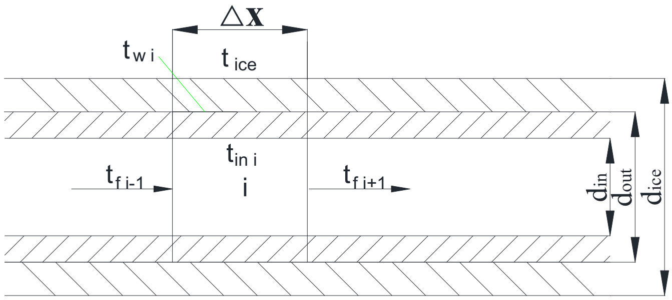

Fig. 7 Schematic diagram of the heat transfer in icy conditions

- Convective heat transfer between the seawater and the ice layer Q_i:

$$Q_i = h_{ice} \cdot \pi \cdot d_{icei} \cdot \Delta x \cdot (t_{sea} - t_{icei}) \tag{1}$$

- Conduction heat transfer through the ice layer Q_i:

$$Q_i = 2\pi \cdot \lambda_{ice} \cdot \Delta x \cdot \frac{t_{icei} - t_{wi}}{\ln \dfrac{d_{icei}}{d_{out}}} \tag{2}$$

Where λ_{ice} is the heat conductivity coefficient of the ice layer and equals 2.03 W/(m·K);

- Conduction heat transfer through the pipe wall Q_i:

$$Q_i = 2\pi \cdot \lambda_w \cdot \Delta x \cdot \frac{t_{wi} - t_{ini}}{\ln \dfrac{d_{out}}{d_{in}}} \tag{3}$$

- Convection heat transfer between the intermediate medium and the inner wall of the pipes Q_i:

$$Q_i = h_{in} \cdot \pi \cdot d_{in} \cdot \Delta x \cdot (t_{ini} - t_{fi}) \tag{4}$$

The internal convective heat transfer coefficient is proposed by Manlapaz and Churchill [7]:

$$Nu_{in} = [\left(\frac{48}{11} + \frac{51/11}{\left(1 + 1342/PrHe^2\right)^2} \right)^3 + 1.816\left(\frac{He}{1 + \dfrac{1.15}{Pr}}\right)^{3/2}]^{1/3} \tag{5}$$

$$Re_{in} = \frac{u_{in} d_{in}}{\gamma} \tag{6}$$

$$Pr_{in} = \frac{\gamma}{a} \tag{7}$$

The critical Reynold number can be calculated by the equation fitted by Ito [8]:

$$\mathrm{Re}_{cr} = 20000\left(\frac{d_{in}}{D}\right)^{0.32} \tag{8}$$

The critical Reynold numbers of the inside circles and outside circles are 7384 and 6564, respectively. The Reynold numbers of the inner tubes is 3721 while the flow rate of the intermediate medium is 0.4m/s. Therefore, the flow of intermediate medium inner the heat exchanger in this study is laminar flow.

- Heat transfer of the intermediate medium Q_i:

$$Q_i = \frac{\pi}{4} \cdot d_{in}{}^2 \cdot u_{in} \cdot \rho_{in} \cdot C_{in} \cdot \frac{(t_{fi+1} - t_{fi-1})}{2} \tag{9}$$

The external convective heat transfer coefficient can be calculated by the tested temperature distribution and average ice thickness every 10m. The equations must be solved iteratively. At the end, the external convective heat transfer coefficients every 10 m can be calculated. The average external convective heat transfer coefficients are varied from 447.8 to 762.5 W/m2·K while the seawater flow rate changes from 0.12 to 0.34m/s. For the instability of seawater flow rate, the external convective heat transfer coefficients can not correspond to the seawater flow rate one-to-one. The minimum external convective heat transfer coefficient is adopted in the mathematical models to ensure the prediction of the model can satisfy the most unfavorable conditions.

The external convective heat transfer coefficients of the heat exchanger were calculated by the experiment results. And then, they were used in the mathematical models. Therefore, the mathematical models could not be validated by the test data. The mathematical models were used to predict the heat transfer process of the the heat exchanger.

In the present study, the external convective heat transfer coefficient used in the simulation program is 447.8 W/m2·K. The equations used to develop the mathematical model are the same with those to solve the external convective heat transfer coefficient. The external convective heat transfer coefficient, the intermediate medium flow rate and inlet temperature are used to calculate the outlet temperature. The equations are solved iteratively and the simulation program is developed based on the VB programming language. Using all the equations above, the temperature distribution along the HCHE in icy condition can be calculated by dividing the whole pipe into small parts in order to calculate fluid inlet and outlet temperatures by using the fluid outlet temperature as fluid inlet temperature of the next part. In the end the temperature profile along the pipe can be calculated.

5. Results and discussion

The heat transfer rate of the helical coil heat exchanger in seawater in icy conditions can be predicted using the models developed. It can be used for the designing of the SWHP systems. Simulations with different parameters are conducted to investigate the effect of variable parameters on heat transfer rate, such as inlet temperature, intermediate medium flow rate, heat exchanger's length and diameter, temperature of seawater. Operations of the system in summer and winter are simulated.

The effects of the inlet temperature of the intermediate medium on the heat transfer rate of the HCHE at different flow rates under icy condition is given in Fig. 8. The results are obtained while the seawater temperature is -1.5°C. It observed that the effect of icing outside the pipes on heat transfer performance is great in the frigid periods in winter. With the decreasing of inlet temperature, the heat transfer grew up. And the ice thickness outside tube gradually accumulated as well, which resulted in the thermal resistance increasing significantly and its increasement was much higher than the growth of heat transfer. As a result, the rate of heat transfer growth was gradually declined as the inlet temperature increased.

The effects of pipe length on heat transfer under different diameter in icy conditions are simulated in this study, as shown in Fig.9. The flow rate of the intermediate medium is 0.4m/s, the inlet temperature is -5°C and the seawater temperature is -1.5°C in icy condition. From Fig.9 we can see that the heat transfer rate increase as the increase of the pipe length and the enlargement of the diameter. While the pipe length is 100m and the diameter is 40mm, the heat transfer rate is almost the same with the one which is 50m long and the diameter is 60mm. Shorten the length of pipes could decrease the heat exchanger's height, which contributes to the steadiness of heat exchanger at the bottom of sea. The enlargement of the diameter was helpful to reduce consumed energy by pump. However, if the diameter is too large, it would be difficult to twine around the fixed mount.

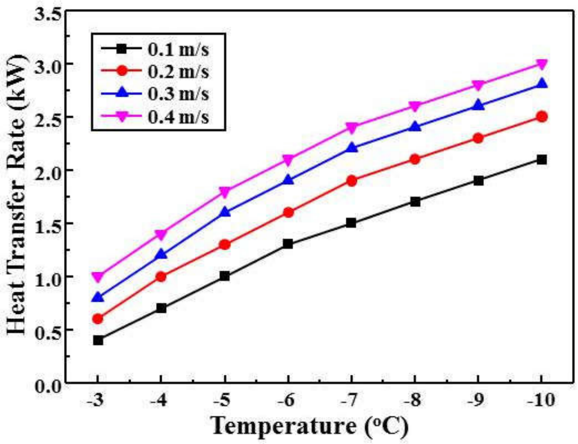

Fig. 8 Effects of inlet temperature of intermediate medium on the heat transfer rate of HCHE at different flow rates under icy conditions in winter

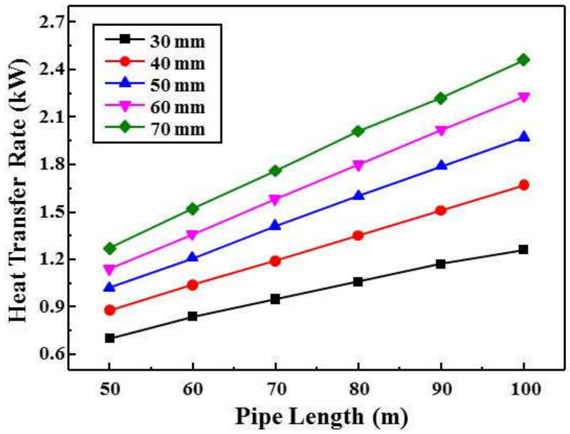

Fig.9 Effect of pipe length on heat transfer in different diameter under icy conditions

Effect of seawater temperature on ice thickness outside the pipes is studied under the condition of inlet temperature is -5 °C and flow rate of intermediate medium is 0.4 m/s. The results are described in Fig.10. With the increase of the pipe length, the ice thickness outside the pipes decreased. From Fig.10 we can see that, the ice thickness varied large at the first 50 m, which means that the heat transfer rate of the HCHE is concentrated in the first 50 m. It can be contributed to the diminishing of the temperature difference between the intermediate medium and the seawater, as the increase of the pipe length. While the seawater temperature is -0.5 °C, there is no ice at the pipe section from 90 m to 100 m. While the seawater temperature is 0 °C, the no ice part begins from 70 m. Thus, with seawater temperature rising, ice thickness and freezing length decreases gradually and the heating quantity get the opposite effect.

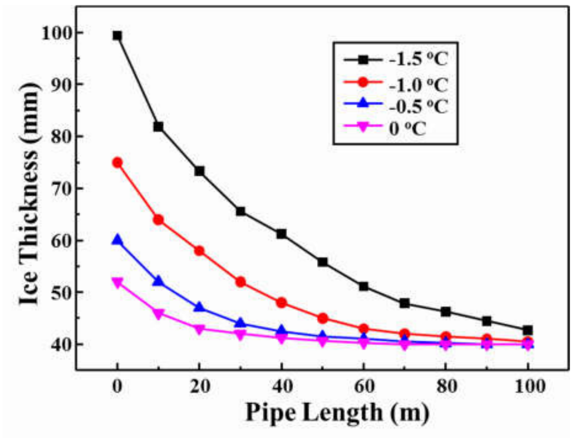

Fig. 10 Effect of seawater temperature on ice thickness outside the pipes

6. Conclusions

The main conclusions of this study can be concluded as:

- In this study, a high-density polyethylene helical coil heat exchanger is firstly adopted by a seawater-source heat pump system and experiments are conducted to study the thermal performance of the heat exchanger in the Bohai Sea, Tianjin, North China.

- The external convective heat transfer coefficients of the HCHE in seawater are experimental studied, which are used to develop the mathematical models of the heat exchanger. While the seawater flow rate varied from 0.12 to 0.34m/s, the average external convective heat transfer coefficients are calculated to be varied from 447.8 to 762.5 $W/m^2{\cdot}K$.

- Mathematical model is developed to simulate the heat transfer process of the heat exchanger in icy condition in winter. The minimum outside heat transfer coefficient is adopted to develop the mathematical model to make sure the prediction of the model can satisfy the most unfavorable conditions.

- Simulations with different parameters are conducted to investigate the effects of variable parameters on heat transfer rate, such as inlet temperature, intermediate medium's flow rate, heat exchanger's length and diameter, temperature of seawater.

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